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A Generalized Flow Correlation for Two-Phase Natural Circulation Loops

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Abstract

Scaling is required in test facilities as full scale testing or experiments are prohibitively expensive or have significant safety implications. The general objective of a scaling analysis is to obtain the physical dimensions and operating conditions of a reduced scale facility capable of simulating the important flow and heat transfer behaviour of the system under investigation. It is essential to know the flow rate to establish the heat transport capability of natural circulation loops. A large number of scaling parameters are available in the literature. But practically it is very difficult to simulate all the given parameters between prototype and model. Another problem associated with the existing scaling laws are that they do not give the steady state flow rate directly whereas most of the proposed dimensionless parameters depend on the flow rate. To overcome these difficulties, a generalized flow correlation has been proposed to simulate the steady state behaviour with just one non-dimensional parameter. The governing equations for homogeneous equilibrium model viz. continuity, momentum and energy equations have been solved for two-phase loops to derive the correlation as

$$\mathrm{Re}_{ss} = C\left[\frac{Gr_m}{N_G}\right]^r$$

To establish the validity and utility of this correlation, a good number of two-phase natural circulation experimental data has been tested with the proposed correlation and found to be in good agreement.

Nomenclature

General symbols

A : flow area, m^2
a : dimensionless flow area, A/A_r
b : constant in equation (7)
D : hydraulic diameter, m
d : dimensionless hydraulic diameter
f : Darcy-Weisbach friction factor
g : gravitational acceleration, m/s^2
Gr_m : modified Grashof number
h : enthalpy, J/kg
H : loop height, m
$\mathcal{H}$: dimensionless enthalpy
K : local pressure loss coefficient
l : dimensionless length, L_i/L_t
L : length, m
N : total number of pipe segments
N_G : dimensionless parameter (Geometric contribution of loop to friction number)
p : constant in equation (7)
P : pressure, N/m^2
q'' : heat flux, W/m^2
Q : total heat input rate, W
Re : Reynolds number, $DW/A\mu$
s : co-ordinate around the loop, m
S : dimensionless co-ordinate around the loop, s/H
T : temperature, K
v : specific volume, m^3/kg
V_t : total loop volume, m^3
W : mass flow rate, kg/s
x : quality
z : elevation, m
Z : dimensionless elevation, z/H

Greek Symbols

α : void fraction
β_{tp} : two-phase thermal expansion coefficient, kg/J
μ : dynamic viscosity, $N\,s/m^2$

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ϕ_{LO}^2 : two-phase friction multiplier

$\bar{\phi}_{LO}^2$: average two-phase friction multiplier

ρ : density, kg/m^3

ρ_r : reference density, kg/m^3

ω : dimensionless mass flow rate

Subscripts

c : cooler

eq : equivalent

eff : effective

g : vapor

he : heater exit

i : i^{th} segment

in : inlet

l : liquid

LO : liquid only

m : mean

p : pipe

r : reference value

sp : single phase

ss : steady state

t : total

tp : two-phase

1. Introduction

Two-phase natural circulation is capable of generating larger buoyancy forces and hence flows. Two-phase natural circulation finds application in nuclear steam generators, thermosyphon boilers, boilers in fossil fuelled power plants, reactor core cooling etc. The heat transport capabilities of natural circulation loops depend on the flow rate it can generate. For two-phase natural circulation loops, explicit correlations for steady state flow are not available. This makes it difficult to compare the performance of different two-phase natural circulation loops. Therefore, we present an analytical correlation for steady state flow, which is then non-dimensionalized to obtain a generalized correlation. This generalized correlation is then tested against data generated in five test facilities differing in diameter.

Pioneering work in the field of scaling laws for nuclear reactor systems have been carried out by Nahavandi et al. [7], Zuber [16], Ishii-Kataoka [4], Kocamustafaogullari-Ishii [5], Schwartzbeck et al. [11], Yadigaroglu et al. [15], Reyes Jr. [10] and Vijayan et al. [13]. The scaling law proposed by Zuber is also known as the power-to-volume scaling philosophy. The integral test facility being set-up to simulate the Advanced Heavy Water Reactor (AHWR) has been designed based on this philosophy. However, the power-to-volume scaling philosophy has certain inherent distortions (especially in downsized components), which can suppress certain natural circulation specific phenomena like the instability (Nayak et al. [8]). Scaling laws provided by Ishii-Kataoka [4] had been widely used for two-phase natural circulation loops. The PUMA facility simulating the SBWR has been designed based on this philosophy. Kocamustafaogullari-Ishii [5] has given a scaling law for two-phase flow transients using reduced pressure Freon (R-11 or R-113) systems. A flow pattern transition dependent scaling law has been given by Schwartzbeck et al. [11]. Yadigaroglu et al. [15] had given a fluid-to-fluid scaling law for a gravity and flashing driven natural circulation loop. Reyes Jr. [10] has applied catastrophe functions to describe the scaling for two-phase natural circulation loops. One of the problems associated with these scaling laws is that the number of similarity groups are too many and they do not provide steady state or stability solutions in terms of the proposed similarity groups. Therefore, testing of these scaling laws with the available experimental data is rather difficult without the use of system codes. This arises due to the fact that more than one scaling parameter is a function of the flow rate, which for a natural circulation loop is not known apriori.

To overcome this problem, Vijayan et al. [14] proposed a scaling procedure by which the steady state flow rate can be obtained as a function of just one similarity group for uniform diameter loops with adiabatic pipes operating without any sub cooling. But the proposed correlation had not been tested rigorously. In the present paper, a generalized scaling philosophy has been proposed for two-phase natural circulation loops. This has been derived in the same line as that of Vijayan et al. [14]. The similarity parameter has been tested against the available data on steady state flow. This exercise has shown that the steady state behaviour of two-phase natural circulation loops can be simulated by a single dimensionless parameter.

2. Steady State Behaviour of Two-Phase Natural Circulation Loops

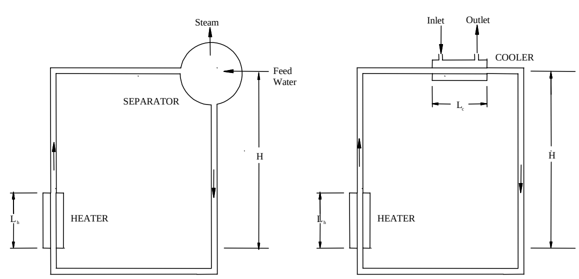


Fig. 1: Schematic of uniform diameter natural circulation loops.

2.1 Governing Equations

The one-dimensional steady state Navier-Stokes equations for two-phase natural circulation system can be written as follows:

Continuity equation:

$$\frac{d}{ds}\left(\frac{W}{A}\right)=0 \qquad (1)$$

Energy equation:

$$\frac{W}{A}\frac{dh}{ds} = \begin{cases} \dfrac{4q''_h}{D_h} \\ 0 \\ -\dfrac{4q''_c}{D_c} \end{cases} \quad (2)$$

Momentum equation:

$$\frac{W^2}{A^2}\frac{d}{ds}\left(\frac{1}{\rho}\right) = -\frac{dP}{ds} - \rho g \sin\theta - \frac{fW^2}{2D\rho A^2} - \frac{KW^2}{2\rho A^2 L_t} \quad (3)$$

In the energy equation uniform heat flux is assumed. Noting that $v = 1/\rho$ and integrating the momentum equation around the circulation loop

$$\frac{W^2}{A^2}\oint dv = -\oint dP - g\oint \rho\, dz - \frac{fW^2 L_t}{2D\rho A^2} - \frac{KW^2}{2\rho A^2} \quad (4)$$

Where $dz = ds.\sin\theta$

Noting that $\oint dv = \oint dP = 0$ for a closed loop, we can write

$$0 = -g\oint \rho\, dz - \frac{fW^2 L_t}{2D\rho A^2} - \frac{KW^2}{2\rho A^2} \quad (5)$$

In the two-phase regions, the density is assumed to vary as $\rho_{tp} = \rho_r\left[1 - \beta_{tp}(h - h_r)\right]$ in the buoyancy force term. For the estimation of frictional pressure loss, liquid density ρ_l is used in single-phase regions and the two-phase density ρ_{tp} is used in the riser. For the heater an average density ρ_m is used. With these and the two-phase friction factor multiplier ϕ_{LO}^2, equation (5) can be rewritten as

$$0 = g\rho_r\beta_{tp}\oint h\, dz - \left[\sum_{i=1}^{N_{sp}}\left(\frac{fL_{eff}}{D}\right)_{i,sp}\frac{W_i^2}{\rho_l A_i^2} + \bar{\phi}_{LO}^2\sum_{i=N_{sp}}^{N_{he}}\left(\frac{fL_{eff}}{D}\right)_{i,sp}\frac{W_i^2}{\rho_l A_i^2} + \phi_{LO}^2\sum_{i=N_{he}}^{N_t}\left(\frac{fL_{eff}}{D}\right)_{i,sp}\frac{W_i^2}{\rho_l A_i^2}\right] \quad (6)$$

Now the above equations can be non-dimensionalized using the following substitutions:

$$\omega = \frac{W}{W_{ss}},\ \mathcal{H} = \frac{h - h_r}{(\Delta h)_{ss}},\ Z = \frac{z}{H},\ S = \frac{s}{H},\ a_i = \frac{A_i}{A_r},\ d_i = \frac{D_i}{D_r},$$

$$l_i = \frac{L_i}{L_t},\ A_r = \frac{\sum_{i=1}^{N} A_i L_i}{\sum L_i} = \frac{V_t}{L_t},\ D_r = \frac{\sum_{i=1}^{N} D_i L_i}{L_t},\ (l_{eff})_i = \frac{(L_{eff})_i}{L_t}$$

$$\rho_r = \rho_{in},\ h_r = h_{in},\ f_i = \frac{p}{\mathrm{Re}_i^b} = \frac{p}{\mathrm{Re}_{ss}^b}\frac{\omega^{-b} a_i^b \mu_i^b}{d_i^b \mu_r^b} \quad (7)$$

$\mathrm{Re}_{ss} = \dfrac{D_r W_{ss}}{A_r \mu_r}$, $L_{eff} = L_i + L_{eq}$ and

$$\mu_r = \frac{\sum_i \mu_i L_i}{\sum_i L_i}$$

At steady state putting $\omega = \dfrac{W}{W_{ss}} = 1$, $\mu_i = \mu_r$ and $q_c = q_h$ the non-dimensional equations will become

$$\frac{d}{dS}\left(\frac{\omega}{a}\right) = 0 \quad (8)$$

$$0 = \frac{g\rho_r\beta_{tp}H(\Delta h)_{ss}A_r V_t \rho_r}{L_t W_{ss}^2}\oint \mathcal{H}\, dZ - \frac{p}{2}\frac{\mathrm{Re}_{ss}^{2-b}\mu_r^2}{D_r^2\rho_l}\frac{A_r V_t\rho_r}{L_t W_{ss}^2}N_G \quad (9)$$

where

$$N_G = \frac{L_t}{D_r}\left[\sum_{i=1}^{N_{sp}}\frac{(l_{eff})_i}{d_i^{1+b}a_i^{2-b}} + \bar{\phi}_{LO}^2\sum_{i=N_{sp}}^{N_{he}}\frac{(l_{eff})_i}{d_i^{1+b}a_i^{2-b}} + \phi_{LO}^2\sum_{i=N_{he}}^{N_t}\frac{(l_{eff})_i}{d_i^{1+b}a_i^{2-b}}\right] \quad (10)$$

It may be noted that for uniform diameter loop, N_G reduces to the following equation

$$N_G = \frac{L_t}{D_r}\left[(l_{eff})_{sp} + \bar{\phi}_{LO}^2(l_{eff})_{sp}^{he} + \phi_{LO}^2(l_{eff})_{he}^t\right] \quad (11)$$

$$\frac{d\mathcal{H}}{dS} = \phi_h\frac{V_t}{V_h} \quad \text{Where } \phi_h = a_h\frac{H}{L_t} \quad (12)$$

$$\frac{d\mathcal{H}}{dS} = -\phi_c\frac{V_t}{V_c} \quad \text{Where } \phi_c = a_c\frac{H}{L_t} \quad (13)$$

After applying the proper boundary conditions for the heater and cooler section, it can be shown that $\oint \mathcal{H}\, dZ = 1$. Hence,

$$W_{ss} = \left[\frac{2}{p}\frac{g\rho_r\beta_{tp}HQD_r^b A_r^{2-b}\rho_l}{\mu_r^b N_G}\right]^{\frac{1}{3-b}} \quad (14)$$

$$\mathrm{Re}_{ss} = 0.176776\left(\frac{Gr_m}{N_G}\right)^{0.5} \quad \text{Laminar flow} \quad (15)$$

$$\mathrm{Re}_{ss} = 1.9561\left(\frac{Gr_m}{N_G}\right)^{0.36364} \quad \text{Turbulent flow} \quad (16)$$

Where $Gr_m = \dfrac{D_r^3\rho_r\rho_l\beta_{tp}gHQ}{A_r\mu_r^3}$

2.2 Estimation of β_{tp}

We have proposed a new parameter, β_{tp}, which is the two-phase thermal expansion coefficient. We have assumed a linear variation of density inside the heater. Hence, to check the accuracy of this assumption the density has been calculated for various pressure and quality. It was found that beyond a quality of about 0.1 (10%), the two-phase thermal expansion coefficient is practically a constant for all pressures and its value is the same, independent of pressure and quality as shown in Fig. 2. β_{tp} in terms of densities can be calculated using the relation

$$\beta_{tp} = \frac{1}{v}\left(\frac{\partial v}{\partial h}\right) = \frac{1}{\frac{v_{in}+v_{exit}}{2}}\left(\frac{v_{exit}-v_{in}}{h_{exit}-h_{in}}\right) = \frac{\rho_{in}-\rho_{exit}}{\left(\frac{\rho_{exit}+\rho_{in}}{2}\right)\Delta h} \qquad (17)$$

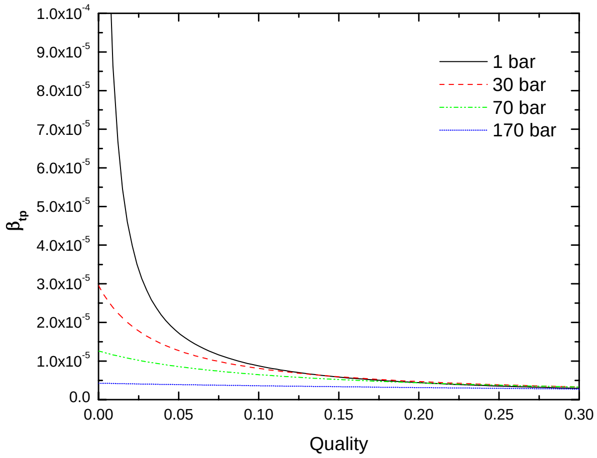


Fig. 2: Variation of β_{tp} with pressure and quality.

2.3 Estimation of ϕ_{LO}^2

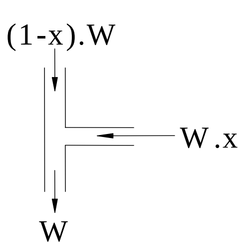
$$\phi_{LO}^2 = \frac{\rho_l}{\rho_{exit}}\left[\frac{1}{1+x\left(\frac{\mu_l}{\mu_g}-1\right)}\right]^b$$

and $$\bar{\phi}_{LO}^2 = \frac{\rho_l}{\bar{\rho}_{exit}}\left[\frac{1}{1+\frac{x}{2}\left(\frac{\mu_l}{\mu_g}-1\right)}\right]^b \qquad (18)$$

where

$$\rho_{exit} = \frac{\rho_g \rho_l}{x(\rho_l-\rho_g)+\rho_g}, \quad \bar{\rho}_{exit} = \frac{\rho_g \rho_l}{x/2\,(\rho_l-\rho_g)+\rho_g}$$

2.4 Estimation of h_{in}



At the mixing section of the SD

$$W(1-x)h_l + W_{feed}\, h_{feed} = W\, h_{in}$$

$$W(1-x)h_l + W\, x\, h_{feed} = W\, h_{in} \quad (\text{Since } W_{feed} = W\, x)$$

$$h_{in} = h_l + x\left(h_{feed} - h_l\right) \qquad (19)$$

2.5 Estimation of ρ_{in}

$$W(1-x)\, c_p\, T_{sat} + W\, x\, c_p\, T_{feed} = W\, c_p\, T_{in}$$

$$T_{in} = T_{sat} + x\left(T_{feed} - T_{sat}\right)$$

Now knowing the system pressure, P, and the inlet temperature, T_{in}, the inlet density ρ_{in} can be calculated.

3. Experimental Validation

3.1 Experimental Loop

To validate the above proposition, an experimental facility was constructed with the length dimensions as in Fig. 3. The experiments were conducted for three different diameters of pipe namely 10.21 mm (½”), 15.74 mm (¾”) and 19.86 mm (1”) respectively. For all the different loop diameters, the steam drum, the condenser and the associated piping (the portion inside the rectangular box in Fig. 3) were the same. The steam drum was made up of 59 mm inside diameter (2.5” NB Sch 80) pipe. The loop was designed for a pressure 125 bar and temperature of 400 °C with maximum operating power as 10 kW. The vertical heater section was direct electrically heated. The steam so produced was condensed in the condenser and the condensate was returned to the steam drum. The loop was extensively instrumented to measure temperature, pressure, differential pressure, level, flow rate, void fraction and its distribution. The void fraction was measured using both Neutron Radiography (NRG) and Conductance Probe (CP) techniques. Further details of the loop are available in the report by Dubey et al. [1].

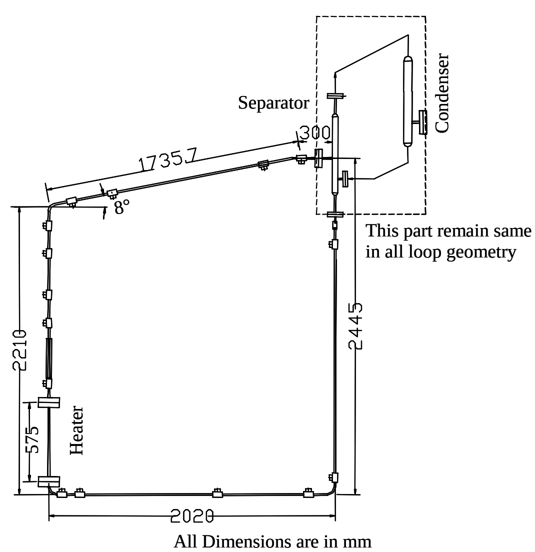


Fig. 3: Experimental loop.

3.2 High Pressure Natural Circulation Loop

In addition, experimental data were generated in a 2" loop shown in Fig. 4. In this facility, experiments were carried out for power ranging from 0-40 kW and pressure 1-70 bar. Further details of the facility are available in Naveen et al. [9].

The elevation of the primary loop is about 3.3 m and the length of heating section is about 1.25 m. The important design parameters of the loop are:
Design pressure = 114 kg/cm^2
Design temperature = 315 ^oC
The inside diameter of different components of the loop are as given below:

Component	Pipe	I.D (in mm)
Test Section	50 mm NB Sch. 40	52.5
Loop	50 mm NB Sch. 80	49.25
Steam Drum	150 mm NB Sch. 120	139.7

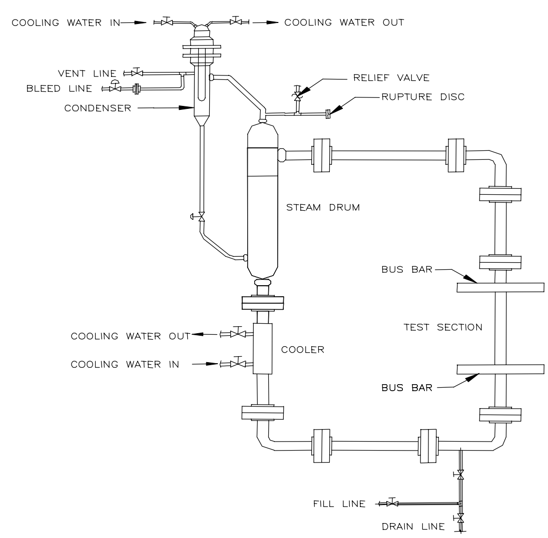


Fig. 4: High pressure natural circulation loop.

3.3 Bettis Natural Circulation Loop (Mendler et al.)

Figure 5 shows the heated test section and natural circulation loop at Bettis Atomic Power Laboratory, Pittsburgh, USA. The main loop piping was fabricated from Sch 80 type SS 304, and was in the shape of a vertical rectangle 4.4323 m (14.5 ft) high and 4.5466 m (15 ft) long. Heat was added uniformly to the lower part of the left vertical leg through an electrically heated rectangular channel test section. The test section was connected to a riser made from 50.8 mm (2") pipe; the other vertical leg is the down comer and was made from 38.1 mm (1 ½") pipe. The top horizontal leg consisted of a double pipe heat exchanger. The bottom horizontal leg contained a 8.636 mm (0.340") diameter orifice and a preheater. The rectangular test sections were 685.8 mm (27") long and 25.4 mm (1") wide and were fabricated of SS 304. Here, 2.54 mm (0.1") nominal spacing was taken as the natural circulation data were available for this dimension only. Further details of the loop can be obtained from Mendler et al. [6].

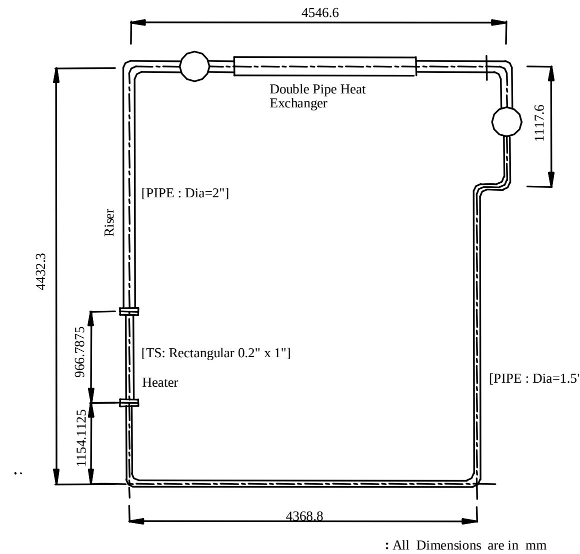


Fig. 5: Bettis natural circulation loop (Mendler et al.).

4. Physical Significance of the Geometrical Parameter (N_G)

The physical significance of N_G can be obtained from the loop pressure drop equation given below

$$\Delta P_t = RW^2 / (2\rho_r)$$

where the total hydraulic resistance, **R** is given by

$$R = \sum_{i=1}^{N} \left(\frac{f_i L_i}{D_i} + K_i \right) \frac{1}{A_i^2} \tag{19}$$

Noting that $(L_{eff})_i = L_i + (L_{eq})_i$, $R = \sum_{i=1}^{N} \left(\frac{fL_{eff}}{DA^2} \right)_i$

Using equation (7) this can be rewritten as

$$R = \frac{L_t}{D_r} \frac{p}{\mathrm{Re}_{ss}^b} \frac{1}{A_r^2} \left[\sum_{i=1}^{N_{sp}} \frac{(l_{eff})_i}{d_i^{1+b} a_i^{2-b}} + \bar{\phi}_{LO}^2 \sum_{i=N_{sp}}^{N_{he}} \frac{(l_{eff})_i}{d_i^{1+b} a_i^{2-b}} + \phi_{LO}^2 \sum_{i=N_{he}}^{N_t} \frac{(l_{eff})_i}{d_i^{1+b} a_i^{2-b}} \right] \text{ for steady state.}$$

From this, using equation (10) we can write

$$RA_r^2 = p \frac{N_G}{\mathrm{Re}_{ss}^b} \;\; or \;\; K_{overall} = N_G \frac{p}{\mathrm{Re}_{ss}^b} \tag{20}$$

where $K_{overall}$ is the effective loss coefficient for the entire loop or the friction number as suggested by Ishii-Kataoka [4]. Equation (20) shows that the friction number can be expressed as the product of two terms, one of which is mainly flow dependent and the other is mainly geometry dependent (except the quality term in

ϕ_{LO}^2). From this, N_G can be considered as the contribution of the loop geometry to the friction number. Again N_G depends upon the nature of the flow (i.e. Laminar or Turbulent, as 'b'is there) and the quality.

5. Comparison of Present Theory with Different Theoretical Models

The mass flow rate calculated for the in-house experimental loop, using the present theory (Equation 14) has been compared with the mass flow rate calculated under the same conditions using RELAP5/ MOD 3.2, TINFLO-S, TINFLO-A and Duffey's Model [2].

Duffey's model is given by:

$$\left(W_{ss}^3\right)_{Duffey} \approx \frac{2\rho_l^2\, g\, Q\, H\, A_r^2\left(\rho_l - \rho_g\right)}{h_{fg}\ \rho_l\ K_{overall}}$$

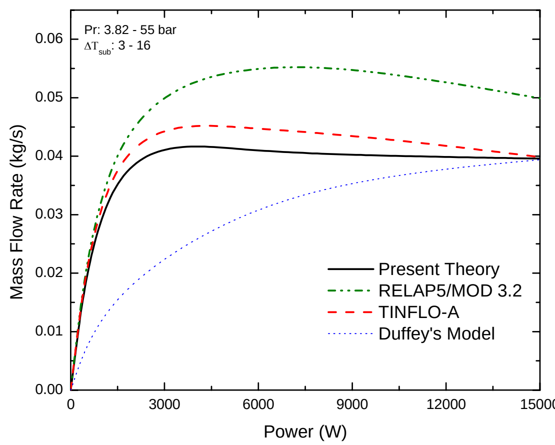


Fig. 6: Variation of flow rate for different pressures and different sub-cooling in ½" Experimental loop.

In TINFLO-S, TINFLO-A and present theory, Blassius friction factor correlation ($f = 0.316\,\mathrm{Re}^{-0.25}$) has been used. The results obtained are shown in Fig. 6 and Fig. 7. As seen from Fig.6, present theory under predicts the mass flow rate as compared to RELAP5/ MOD 3.2. This can be attributed to the fact that RELAP5/MOD 3.2 is based upon a two-fluid model where as homogeneous model with β_{tp} and ϕ_{LO}^2 estimated by equation (17) and (18) respectively, is used in the present theory. Closer agreement could be obtained with other models for ϕ_{LO}^2.

6. Testing of the Steady State Correlation with Experimental Data

The steady state data from five different two-phase natural circulation loops are compared with the theoretical correlation in Fig. 8. The experimental data is observed to be very close to the theoretical correlation (within an error bound of +/- 40%) for all the two-phase natural circulation loops confirming the validity of the correlations given in equation (16). The data of all the loops fall in the parameter range given in Table-1.

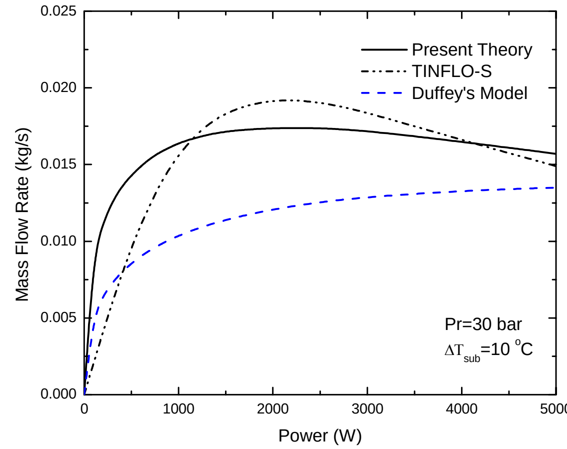


Fig. 7: Variation of flow rate for constant pressure and constant sub-cooling in ½" Experimental loop.

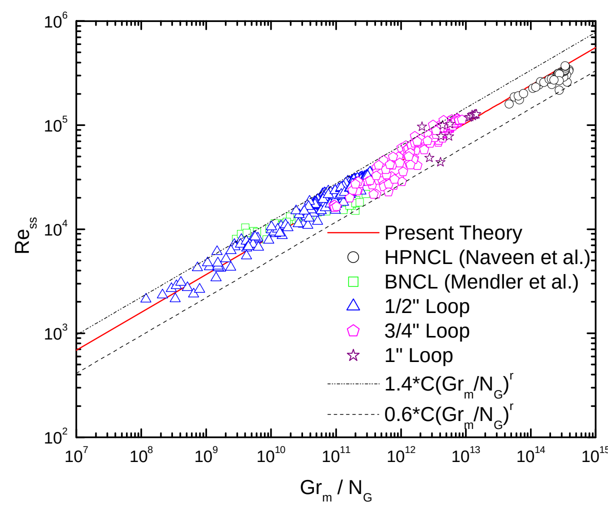


Fig. 8: Comparison of present theory and experimental results.

7. Error Analysis

An error analysis was carried out by standard statistical procedure. The error (e_i), mean error (e_m), mean of absolute error (e_{ma}), root mean square error (e_{rms}) and standard deviation (σ) are calculated as follows:

$$e_i = \{(\xi_c - \xi_m)/\,\xi_m\} \times 100,\ e_m = \frac{1}{N}\sum_{i=1}^{N} e_i,$$

$$e_{ma} = \frac{1}{N}\sum_{i=1}^{N}\left|e_i\right|$$

$$e_{rms} = \left\{\left(\sum_{i=1}^{N} e_i^2\right)\Big/ N\right\}^{0.5} \ and\ \sigma = \left[\left\{\sum_{i=1}^{N}(e_m - e_i)^2\right\}\Big/ (N-1)\right]^{0.5}$$

where ξ_c and ξ_m are the calculated and measured quantities respectively and N is the total number of data points. The results of error analysis are given in Table 2.

Table 1: Range of parameters for the experimental data

Loops	D_h (mm)	L_t (m)	L_t/D_h	P (bar)	Loop Height, H (m)	x	T_{sub} (^oC)	Power, Q (W)	W_{ss} (kg/s)	Fluid
½" Loop	10.21	8.58	840.42	1-58	2.445	0.008-0.239	0.1-29.0	298.1-5416	0.001-0.0305	Steam-Water
¾" Loop	15.74	8.58	545.15	4-61	2.445	0.004-0.039	0.1-22.0	788 – 7425	0.044-0.1622	Steam-Water
1" Loop	19.86	8.58	432.06	8-59	2.445	0.005-0.011	0.1-13.0	1128-3668	0.108-0.2	Steam-Water
BNCL (Mendler et al.)	8.47	17.8	2100	55-138	4.4323	0.082-0.693	8.0-64.0	8260-64600	0.050-0.10	Steam-Water
HPNCL (Naveen et al.)	52.5	13.4	254.38	2.0-46.0	3.350	0.007-0.017	0.3-2.1	20000-36500	0.9-1.8236	Steam-Water

Table 2: Comparison of various experimental data with present theory

Loops	Mean Error	Mean Absolute Error	R.M.S. Error	Standard Deviation
1/2" Loop	-11.37887198	13.3380579	19.0160694	15.25215313
3/4" Loop	-6.395301032	16.28525846	19.321019	18.28136849
1" Loop	-1.332783139	17.71169874	25.4320005	26.35576819
HPNCL (Naveen et al.)	15.53667704	15.53667704	18.6453846	10.41979251
BNCL(Mendler et al.)	4.562436356	23.12767899	28.2772471	28.31417265

8. Conclusions

A generalized correlation for steady state flow in two-phase natural circulation systems has been presented. For two-phase natural circulation systems, the steady state behaviour can be simulated by preserving Gr_m/N_G same in the model and prototype. The given correlation has been tested with data from five different two-phase natural circulation loops. The experimental results are found to be in reasonable agreement with the proposed correlation.

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